Downloaded from orbit.dtu.dk on: May 05, 2023

DTU Library

Theoretical analysis of the performance of different cooling strategies with the concept of cool exergy

Kazanci, Ongun Berk; Shukuya, Masanori; Olesen, Bjarne W.

Published in:
Building and Environment

Link to article, DOI:
10.1016/j.buildenv.2016.02.013

Publication date:
2016

Document Version
Peer reviewed version

Link back to DTU Orbit

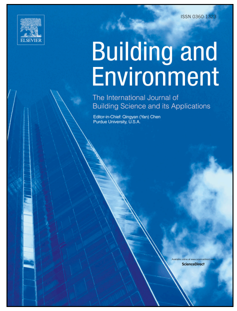
Citation (APA):
Kazanci, O. B., Shukuya, M., & Olesen, B. W. (2016). Theoretical analysis of the performance of different cooling strategies with the concept of cool exergy. *Building and Environment*, *100*, 102-113. https://doi.org/10.1016/j.buildenv.2016.02.013

General rights
Copyright and moral rights for the publications made accessible in the public portal are retained by the authors and/or other copyright owners and it is a condition of accessing publications that users recognise and abide by the legal requirements associated with these rights.

- Users may download and print one copy of any publication from the public portal for the purpose of private study or research.
- You may not further distribute the material or use it for any profit-making activity or commercial gain
- You may freely distribute the URL identifying the publication in the public portal

If you believe that this document breaches copyright please contact us providing details, and we will remove access to the work immediately and investigate your claim.

Accepted Manuscript



Theoretical analysis of the performance of different cooling strategies with the concept of cool exergy

Ongun B. Kazanci, Masanori Shukuya, Bjarne W. Olesen

PII: S0360-1323(16)30050-6

DOI: 10.1016/j.buildenv.2016.02.013

Reference: BAE 4399

To appear in: *Building and Environment*

Received Date: 24 November 2015

Revised Date: 10 February 2016

Accepted Date: 12 February 2016

Please cite this article as: Kazanci OB, Shukuya M, Olesen BW, Theoretical analysis of the performance of different cooling strategies with the concept of cool exergy, *Building and Environment* (2016), doi: 10.1016/j.buildenv.2016.02.013.

This is a PDF file of an unedited manuscript that has been accepted for publication. As a service to our customers we are providing this early version of the manuscript. The manuscript will undergo copyediting, typesetting, and review of the resulting proof before it is published in its final form. Please note that during the production process errors may be discovered which could affect the content, and all legal disclaimers that apply to the journal pertain.

Theoretical analysis of the performance of different cooling strategies with the concept of cool exergy

Ongun B. Kazanci [*,a], Masanori Shukuya [b], Bjarne W. Olesen [a]

[a] International Centre for Indoor Environment and Energy – ICIEE, Department of Civil Engineering, Technical University of Denmark, Nils Koppels Allé, Building 402, 2800 Kgs. Lyngby, Denmark

[b] Department of Restoration Ecology and Built Environment, Tokyo City University, 3-3-1 Ushikubo-nishi, Tsuzuki-ku, Yokohama, 224-8551 Japan

* Corresponding author. Tel.: +4550281327, Fax: +4545932166, e-mail address: onka@byg.dtu.dk

Abstract

The whole chains of exergy flows for different cooling systems were compared. The effects of cooling demand (internal vs. external solar shading), space cooling method (floor cooling vs. air cooling with ventilation system), and the availability of a nearby natural heat sink (intake air for the ventilation system being outdoor air vs. air from the crawl-space, and air-to-water heat pump vs. ground heat exchanger as cooling source) on system exergy performance were investigated.

It is crucial to minimize the cooling demand because it is possible to use a wide range of heat sinks (ground, lake, sea-water, etc.) and indoor terminal units, only with a minimized demand. The water-based floor cooling system performed better than the air-based cooling system; when an air-to-water heat pump was used as the

ACCEPTED MANUSCRIPT

cooling source, the required exergy input was 28% smaller for the floor cooling system. The auxiliary exergy input of air-based systems was significantly larger than the water-based systems.

The use of available cool exergy in the crawl-space resulted in 54% and 29% smaller exergy input to the power plant for the air-based and water-based cooling systems, respectively. For floor cooling, the exergy input to the power plant can be reduced by 90% and 93%, with the use of ground, and use of the ground and the air in the crawl-space, respectively. A new approach to exergy efficiency was introduced and used to prove that the exergy supply from the ground matches well with the low exergy demand of the floor cooling system.

Keywords

radiant floor cooling; air cooling; exergy consumption; ground heat exchanger; air-to-water heat pump; crawl-space

ACCEPTED MANUSCRIPT

1. Introduction

Tightening targets for energy efficiency and energy use reduction in buildings have had significant effects both on residential and non-residential buildings in Europe [1]. The development of passive, low-energy, near zero-energy, and zero-energy buildings has been stimulated by these regulations and environmental concerns, and nearly zero-energy building (nZEB) levels are dictated for new buildings by 2020 in the European Union [1].

The international focus on the residential sector is increasing, and although the energy performance of buildings has increased, issues with the thermal indoor environment and air quality have been reported in low-energy and passive houses [2], [3], [4]. One prominent problem is overheating and it has been reported from Denmark [5], Sweden [3], [6], Finland [4], and Estonia [7]. These findings indicate that cooling in residential buildings is becoming more important and almost a necessity.

Air-based or water-based systems can be used to heat or cool buildings. Although different studies have evaluated the performance of air-based and water-based heating and cooling systems for office buildings [8], [9], [10], and benefits of radiant panel heating and cooling in net zero-energy buildings [11], so far there has only been little focus on residential buildings and dwellings regarding cooling systems and their exergy performance.

In addition to the insights to different systems by energy analyses, exergy analyses articulate more precisely and accurately the different quality of energy sources and flows. “Cool” and “warm” exergy concepts enable us to quantify and to properly account for the “warmth” and “coolness” of a heat source or sink, and exergy flows from these sources and sinks [12], [13], [14].

In this study, the exergy performance of different space cooling systems was compared using a single-family house as a case study. The whole chain of exergy flows were considered from the source until the environment. The effects of cooling demand (studied by means of installing internal vs. external solar shading), space cooling method (floor cooling vs. air cooling with ventilation system) including auxiliary exergy use for pumps and fans, and the availability of a nearby cool exergy source (intake air for the ventilation system being outdoor air vs. air from the crawl-space, and air-to-water heat pump vs. ground heat exchanger as cooling source) on the system performance regarding energy, exergy demand and exergy consumption were studied. The cool exergy concept was used to analyze the crawl-space and the ground.

2. Analyzed space cooling systems

The eight different cooling systems that were studied in this paper are described here, before explaining the exergy calculation method that was used to perform the case studies.

2.1. Determination of the design cooling load

The studied house was assumed to be located in Copenhagen, Denmark. Construction details, description and details of the heating, cooling and ventilation systems of the actual house are given in [15] and [16].

The space cooling load was determined with the assumption of steady-state conditions. The outdoor air temperature was assumed to be 30°C, which is also the environmental (reference) temperature for exergy calculations. For all cases, the indoor temperature was 26°C (air temperature and mean radiant temperature). The relative humidity indoors was assumed to be 55%, resulting in a dew point temperature of 16.3°C.

The house was supported on 30 cm high concrete blocks and this created a crawl-space between the ground and the house’s floor structure. When the intake air was taken from the crawl-space, the fresh air temperature coming into the air handling unit (AHU) or to the indoor space was 21.3°C, due to the pre-cooling of the outdoor air by the ground surface under the crawl-space.

The internal heat gain was assumed to be 4.5 W/m^2 which represents two persons at 1.2 met and other household equipment. For the floor cooling cases, a ventilation rate of 0.5 air change per hour (ach) was used to provide fresh air to the indoors [17]. For the air cooling cases, the supply air flow rate was calculated based on the cooling load. For all cases, an infiltration rate of 0.2 ach was assumed.

For Copenhagen, Denmark (56° Northern Latitude), in July at noon, assumed direct solar radiation on the South and West directions were 390 and 149 W/m^2, respectively, and the diffuse solar radiation was 32 W/m^2 [18]. The shading coefficients for internal and external solar shading were assumed to be 0.6 and 0.1, respectively (blinds, 45° inclination, light colored) [18].The resulting space cooling loads for different cases are given in Table 2 and Table 3.

2.2. Details of eight cases studied

In order to compare the exergy performance of different cooling systems, the house was assumed to be cooled with a water-based radiant floor cooling system or an air cooling system with the supply of cold air from the air handling unit. The following assumptions were made during the calculation procedure:

- In the actual house, there was a heat exchanger between the radiant system and the heat pump, but for the calculations this heat exchanger was neglected and it was assumed that the water in the floor loops circulated directly through the evaporator of the heat pump. The same was assumed for the air-cooling coil in the AHU.
- The supply air was 100% outdoor air (no recirculation), and the indoor air was assumed to be fully mixed (mixing ventilation).
- It was assumed that there was no heat gain to the floor cooling system, pipes and ducts from the outdoors.

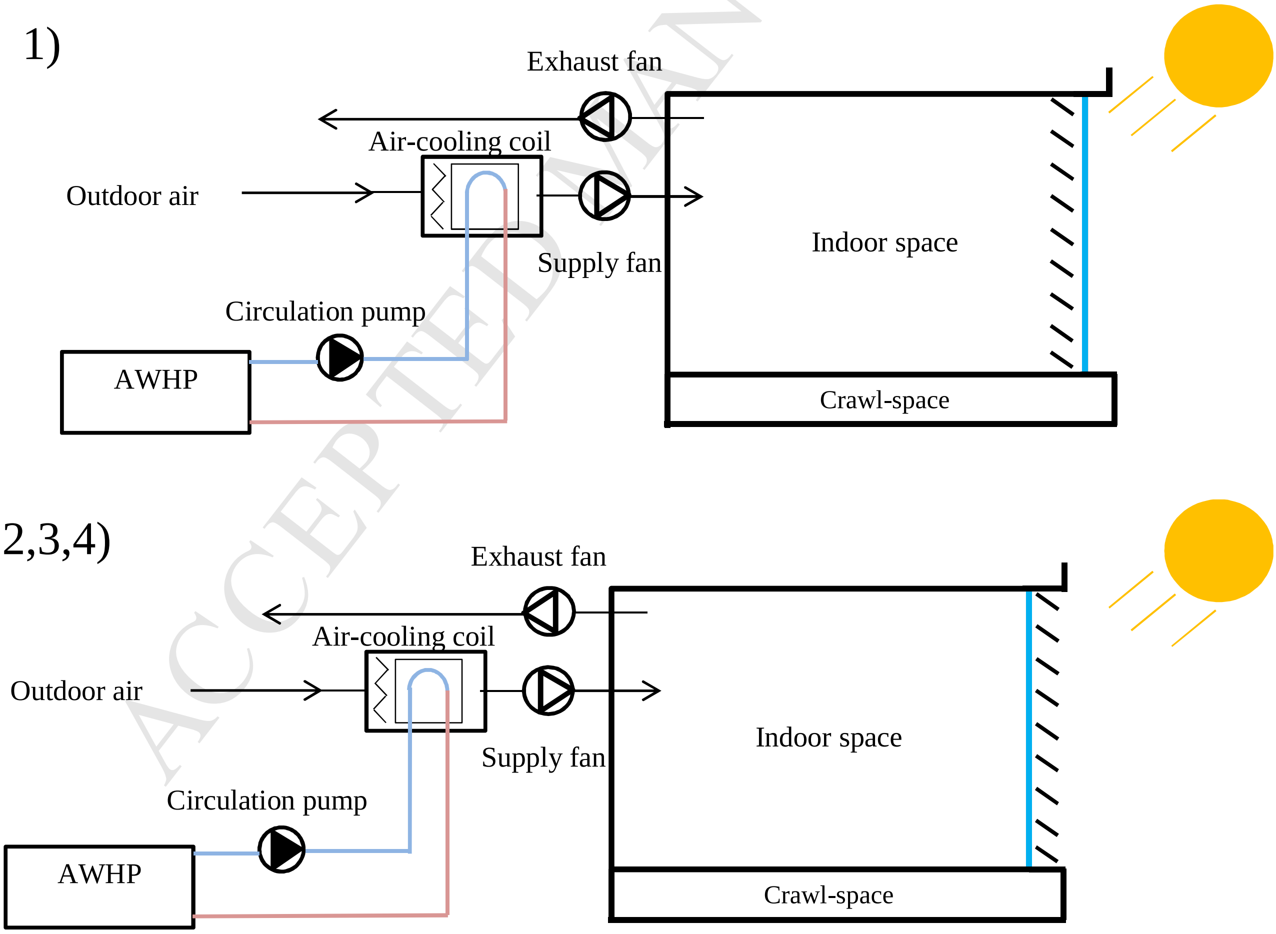
A summary of the investigated cases is given in Table 1, and schematic drawings of the eight cases are given in Fig. 1.

Table 1. Summary of the case studies

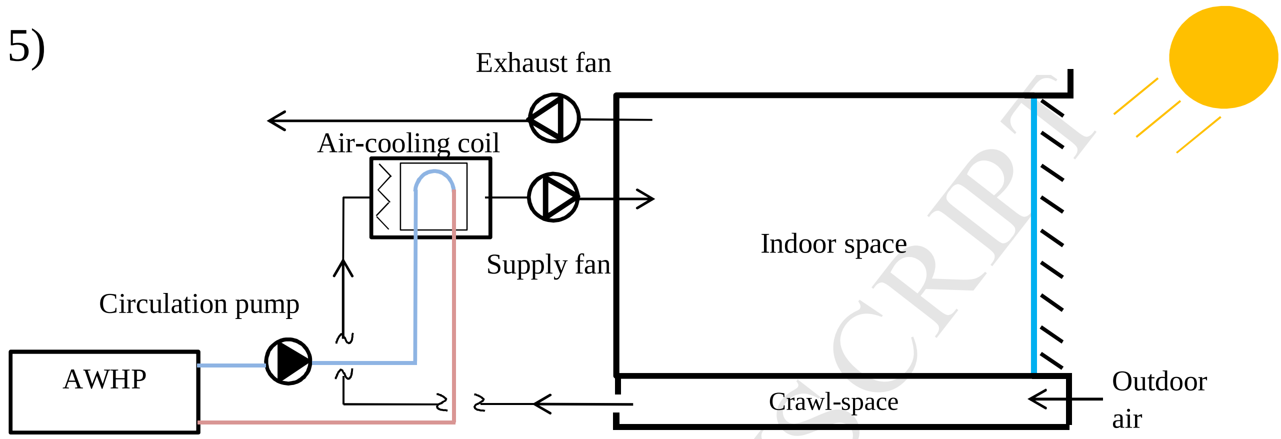
Case	Shading	Cooling	Source	Intake air
1	Internal	AC	AWHP	OA

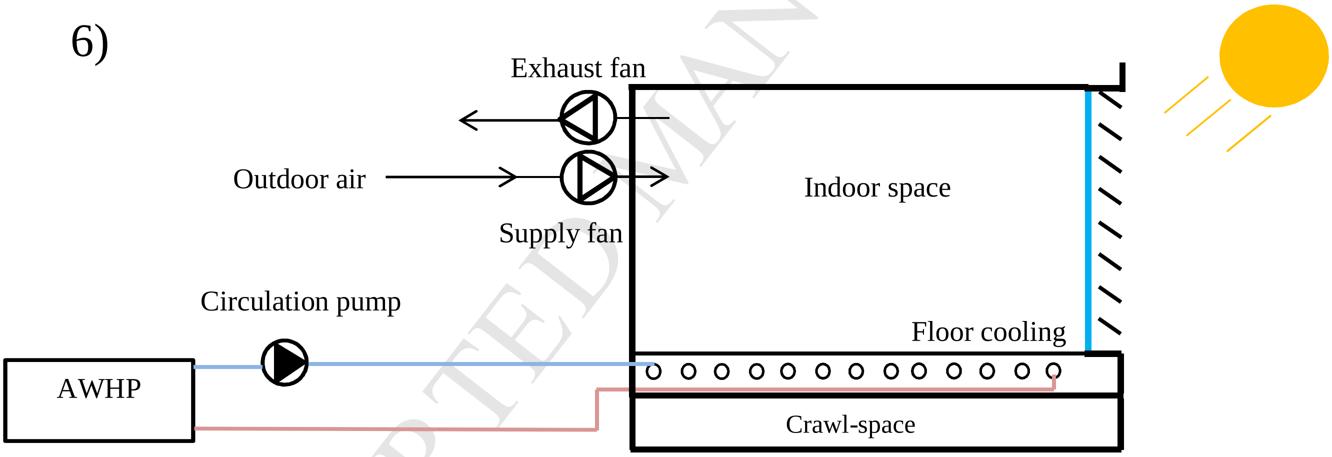
2*	External	AC	AWHP	OA
3*	External	AC	AWHP	OA
4*	External	AC	AWHP	OA
5	External	AC	AWHP	CS
6	External	FC	AWHP	OA
7	External	FC	AWHP	CS
8	External	FC	GHEX	CS

*: Supply air temperatures and air flow rates are different for Cases 2 - 4. Further details are given in Table 3. AC: air cooling, FC: floor cooling, AWHP: air-to-water heat pump, GHEX: ground heat exchanger, OA: outdoor air, CS: crawl-space.



ACCEPTED MANUSCRIPT


5)
Exhaust fan
Air-cooling coil
Supply fan
Indoor space
Circulation pump
AWHP
Crawl-space
Outdoor air


6)
Exhaust fan
Outdoor air
Supply fan
Indoor space
Circulation pump
Floor cooling
AWHP
Crawl-space

ACCEPTED MANUSCRIPT

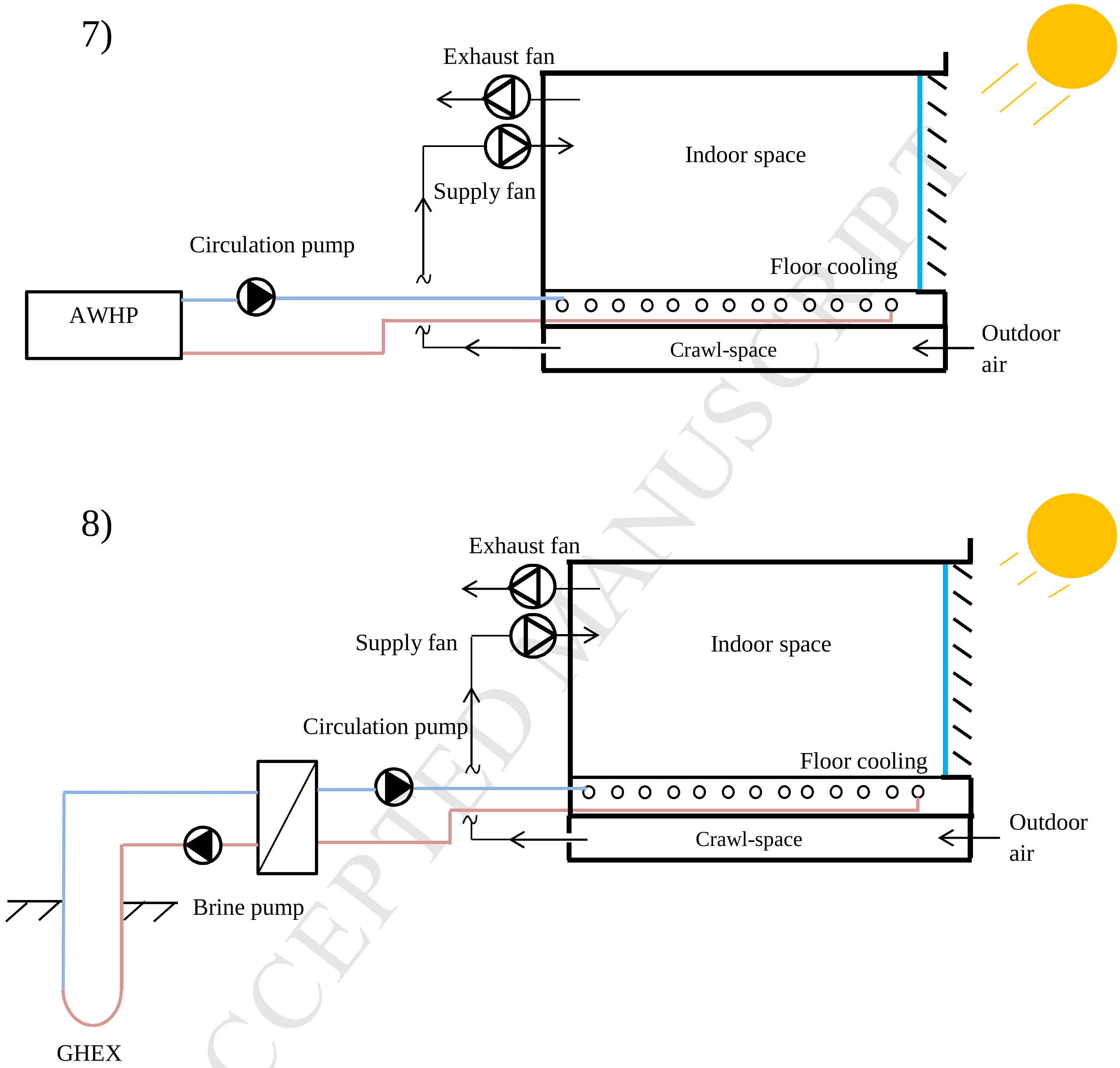


Fig. 1. Schematic drawings of the analyzed cooling systems

2.2.1. Floor cooling cases

For Case 7 and Case 8, the heat to be removed by the floor was 876 W, and for Case 6 it was 1183 W. This corresponds to a cooling load of 19.5 and 26.3 W/m^2-cooled floor area, respectively, and a corresponding average floor surface temperature of 23.2 and 22.2°C. In order to achieve these surface temperatures, the required supply and return water temperatures were 18.6 and 21.6°C for Case 7 and Case 8, and 16.5 and 19.5°C for Case 6. For all cases, the temperature difference between supply and return water flows was assumed to be 3°C. For Case 7 and Case 8, this resulted in a mass flow rate of 250 kg/h, and for Case 6 it was 338 kg/h. A floor covering resistance of 0.05 m^{2}K/W was assumed for all cases to keep the effects of floor covering resistance on the system performance to a minimum [15].

The cooling output, floor surface temperatures and the mass flow rates were calculated according to [19], [20], [21], [22]. The summary of floor cooling cases is given in Table 2.

Table 2. Summary of the floor cooling cases

Case	Space cooling load [W]	Supply and return water temperature [°C]	Cooled floor surface temperature [°C]	Water flow rate [kg/h]
6	1183	16.5 / 19.5	22.2	338
7 & 8	876	18.6 / 21.6	23.2	250

2.2.2. Air cooling cases

The required ventilation rates were calculated based on the space cooling loads and the temperature difference between the supply air and room air temperatures. The water flow rate in the air-cooling coil was calculated based on the heat to be removed from the intake air and the temperature difference in the supply and return water flows to and from the air-cooling coil. The heat to be removed from the intake air corresponds to the required

ACCEPTED MANUSCRIPT

amount of heat to lower the temperature of the intake air to the required supply air temperature, which was 14°C, 17°C or 20°C for respective three cases. The summary of the air cooling cases is given in Table 3.

Table 3. Summary of the air cooling cases (IA: intake air)

Case	Space cooling load [W]	Supply air temperature [°C]	Ventilation rate [ach]	Rate of cooling to IA [W]	Water flow rate in the air-cooling coil [kg/h]
1	3170	14	3.7	4226	725
2	1042	14	1.2	1389	238
3	1042	17	1.6	1505	258
4	1042	20	2.5	1736	298
5	1042	14	1.2	634	109

2.2.3. Air-to-water heat pump, crawl-space, and ground heat exchanger

The temperature of the water leaving the evaporator of the air-to-water heat pump was assumed to be the same as the supply water temperature to the floor loops and to the air-cooling coil.

The coefficient of performance (COP) of the heat pump was obtained from the manufacturer's datasheets as a function of outdoor air temperature and the temperature of the water leaving the evaporator. For floor cooling cases, the required supply water temperature to the floor loops was used to obtain the COP, while for air cooling cases it was assumed that the supply and return water temperatures to and from the air-cooling coil were 7 and 12°C, respectively. The resulting COP values were 3.42 for Case 7, 3.31 for Case 6, and 2.79 for air cooling cases.

In the actual house, the intake air was from the crawl-space and the measurements showed that the air in the crawl-space was warmer than the outdoor air in winter and colder than the outdoor air in summer [16].

Ground heat exchangers can be a good match to couple with high temperature cooling systems [19]. In this study, a single U-tube vertical heat exchanger was assumed to be coupled to the floor cooling system. It was assumed that there was a flat-plate heat exchanger between the floor system and the ground heat exchanger and a brine pump was circulating the anti-freeze mixture consisting of 30% propylene-glycol/water mixture.

The ground temperature of Copenhagen area was taken as 8.3°C [23]. The incoming and outgoing liquid temperatures to and from the borehole were 17 and 13°C, respectively. The corresponding mass flow rate in the borehole was 208 kg/h. It is possible to achieve the necessary cooling of the circulating liquid from 17°C to 13°C at the 40 m depth for the given borehole design. Further details of this ground heat exchanger are given in [23], [24].

Since the temperatures of air in the crawl-space and of the deep ground are different from the outdoor (environmental) temperature, they contain a certain amount of cool (or warm) exergy and they act as immediate cool (or warm) exergy sources.

2.2.4. Fan and pump powers

The power to be supplied to fans and pumps that circulate the heat transfer medium in pipes or in ducts was determined as follows.

The pump power for different cases was obtained from the pump specifications as a function of the water flow rate and the required pressure increase, assuming the pump actually installed in the house.

The fan powers were determined from the measured data at the house. The measurements showed that the AHU was using 67.9 W with a ventilation rate of 0.5 ach (105 m^3/h), which corresponds to a specific fan power (SFP) of 1166 J/m^3 for one fan [16]. This SFP value is in SFP 3 category according to EN 13779:2007 [25]. The fan powers were calculated as a function of the air flow rates, assuming that the fans for the air cooling cases are also in SFP 3 category (1200 J/m^3).

Table 4 summarizes the pump, fan, and their total powers for eight cases studied.

Table 4. Summary of pump, fan, and their total powers for each case

Case	E_{pump} [W]	E_{fans} [W]	E_{total} [W]
1	33.5	528.2	561.7
2	25.0	173.6	198.6
3	25.3	231.5	256.8
4	26.0	347.3	373.3
5	23.0	173.6	196.6
6	26.5	67.9	94.4
7 & 8*	25.2	67.9	93.1

*: The electricity input to the brine pump is not shown in this table, it is not considered as an auxiliary component but rather as a component similar to a heat pump, which is used to deliver the "coolness" from the ground to the floor loops.

3. Basic definitions of exergy and calculation methodology

3.1. Basic definitions

For any system, it is possible to obtain the exergy balance equation from energy and entropy balance equations together with the environmental temperature. In general form, exergy balance equation can be written as follows [13], [26]:

$$[Exergy\ input] - [Exergy\ consumed] = [Exergy\ stored] + [Exergy\ output] \quad (1)$$

where [Exergy consumed] = [Entropy generated] $\cdot$ T_o, and T_o is the environmental (reference) temperature [K], where the system and its components are situated in. The storage term in Eq. (1) disappears for the analyses under steady-state conditions.

ACCEPTED MANUSCRIPT

Eq. (1) indicates that every system consumes a part of the supplied exergy while at the same time the corresponding amount of entropy is generated.

A brief description of "cool" and "warm" exergy concepts are given in Appendix. The following calculations were carried out manually and under steady-state conditions.

3.2. Cooling exergy load

The cooling exergy load is the required rate of exergy to be supplied to the indoor space to maintain the design indoor conditions, and it can be defined as

$$X_{cooling} = -Q_{cooling}\left(1 - \frac{T_o}{T_i}\right) \geq 0 \tag{2}$$

where $X_{cooling}$ is the cooling exergy load [W], $Q_{cooling}$ is the space cooling energy load [W], and T_i is indoor temperature (air and mean radiant temperatures) [K].

3.3. Exergy supplied to the indoor space

The exergy supplied to the indoor space through floor cooling and through the supply air can be calculated using Eqs. (3) and (4), respectively.

$$X_{FC,out} = -Q_{cooling}\,(1 - \frac{T_o}{T_{S,FC}}) \geq 0 \tag{3}$$

$$\Delta X_{AC,out} = V_{sa} c_a \rho_a \left\{(T_{sa} - T_i) - T_o \ln\frac{T_{sa}}{T_i}\right\} \tag{4}$$

where $X_{FC,out}$ is the exergy supplied from floor cooling system to the indoor space [W], $T_{S,FC}$ is the average temperature of the cooled floor surface [K], $\Delta X_{AC,out}$ is the net exergy supplied by cold air to the indoor space (the difference in the amount of exergy carried between the supply air and the indoor air) [W], V_{sa} is the volumetric flow rate of supply air [m^3/s], c_a is the specific heat capacity of air [J/kgK], ρ_a is the density of air [kg/m^3], and T_{sa} is the supply air temperature [K].

The exergy consumed within the indoor space can be obtained as the difference between the exergy supplied to the indoor space and the cooling exergy load.

3.4. Exergy input, output and consumption in the ground, flat-plate heat exchanger, floor, and air-cooling coil

In order to get a complete understanding of the exergy flows in the whole cooling system, it is necessary to start from the ground and to identify the exergy consumption processes. The net exergy input to the circulating anti-freeze mixture from the ground, ΔX_{ground} [W], is obtained from the following equation.

$$X_g - X_{c,ground} = \Delta X_{ground} \tag{5}$$

where $X_g = -Q_g(1 - \frac{T_o}{T_g})$ (6)

$$\Delta X_{ground} = V_g c_{pgw} \rho_{pgw} \left\{ \left(T_{g,out} - T_{g,in}\right) - T_o \ln \frac{T_{g,out}}{T_{g,in}} \right\} \tag{7}$$

X_g is the "cool" exergy flow rate from the ground to the anti-freeze mixture [W], $X_{c,ground}$ is the exergy consumption rate in the ground [W], Q_g is the rate of heat removed from the anti-freeze mixture to the ground [W], T_g is the ground temperature [K], V_g is the volumetric flow rate in the U-tube heat exchanger in the ground [m^3/s], c_{pgw} is the specific heat capacity of the anti-freeze mixture [J/kgK], ρ_{pgw} is the density of the anti-freeze mixture [kg/m^3], $T_{g,out}$ is the temperature of the anti-freeze mixture going out from the ground [K], and $T_{g,in}$ is the temperature of the anti-freeze mixture going into the ground [K].

The exergy consumption in the flat-plate heat exchanger, $X_{c,HEX}$ [W], is obtained from the exergy balance equation, Eq. (8).

$$\Delta X_{ground} - X_{c,HEX} = \Delta X_w \tag{8}$$

where $\Delta X_w = X_{w,supply} - X_{w,return}$ (9)

ΔX_w is the net exergy input from the supply and return water [W], $X_{w,supply}$ is the rate of exergy carried by the supply water into the floor [W], $X_{w,return}$ is the rate of exergy carried by the return water from the floor [W].

The values of $X_{w,supply}$ and $X_{w,return}$ are calculated from the following equation.

$$X_w = V_w c_w \rho_w \left\{ (T_w - T_o) - T_o \ln \frac{T_w}{T_o} \right\} \tag{10}$$

where V_w is the volumetric flow rate of water [m^3/s], c_w is the specific heat capacity of water [J/kgK], ρ_w is the density of water [kg/m^3], and T_w is the supply or return water temperature [K].

The exergy consumption rate in the floor structure, $X_{c,floor}$ [W], is calculated from the following exergy balance equation.

$$\Delta X_W - X_{c,floor} = X_{FC,out} \tag{11}$$

where ΔX_w and $X_{FC,out}$ are given by Eqs. (9) and (3), respectively.

The exergy consumption in the air-cooling coil in the AHU, $X_{c,coil}$ [W], is obtained from

$$\Delta X_{w,coil} - X_{c,coil} = \Delta X_a \tag{12}$$

$$\Delta X_{w,coil} = X_{w,supply,coil} - X_{w,return,coil} \tag{13}$$

$$\Delta X_a = X_{a,out} - X_{a,in} \tag{14}$$

where $\Delta X_{w,coil}$ is the net exergy input by the water to the air-cooling coil [W], $X_{w,supply,coil}$ is the rate of exergy carried by the water entering the air-cooling coil (from the heat pump) [W], $X_{w,return,coil}$ is the rate of exergy carried by the water leaving the air-cooling coil (to the heat pump) [W], $X_{a,out}$ is the rate of exergy carried by the air leaving the air-cooling coil [W], and $X_{a,in}$ is the rate of exergy carried by the air entering the air-cooling coil [W].

Eq. (10) can be applied to the calculation of $X_{w,supply,coil}$ and $X_{w,return,coil}$. In the case of air instead of water, Eq. (10) is also applied with the replacement of the values of volumetric flow rate, specific heat capacity, density and respective temperatures from water to air. Eq. (10) is also used to calculate the rate of cool or warm exergy carried by the air flowing in from the crawl-space.

3.5. Exergy input to the power plant

It was assumed that the electric power supplied to the heat pump, pumps, and fans was generated in a remote, natural gas fired power plant. The exergy input required to the power plant can be determined from

$$E_{HP} = \frac{Q_{HP,cooling}}{COP} \tag{15}$$

$$X_{in,power\ plant} = \frac{E_{HP}}{\eta_{TOT}} r \tag{16}$$

where E_{HP} is power (electricity) input to the heat pump [W], $Q_{HP,cooling}$ is the rate of heat to be removed by the heat pump to the water circulating through the evaporator [W], COP is the coefficient of performance, $X_{in,\ power\ plant}$ is the exergy input to the power plant through natural gas [W], η_{TOT} is the total efficiency including conversion efficiency of the power plant, distribution and transmission efficiencies of the grid (assumed to be 0.35 [13]), and r is the ratio of chemical exergy to higher heating value of natural gas (assumed to be 0.93 [13]).

For the value of $Q_{HP,cooling}$, the space cooling energy load is used in the floor cooling cases, and the rate of heat to be removed from the intake air is used in the air cooling cases.

Exergy input required at the power plant for the pump and fans are calculated using Eq. (16) by replacing the E_{HP} with respective pump power (E_{pump}) and fan power (E_{fans}).

3.6. Exergy efficiency

One way of evaluating the exergy performance of cooling systems is to use exergy efficiency. Conventional definition of the exergy efficiency can be used but it may fail to capture the effects of exergy supply to the cooling system from the immediate natural exergy sources, such as the ground and the crawl-space. Therefore, three kinds of exergy efficiency as defined in Eqs. (17) - (19) were used:

$$\eta_{x,conventional} = \frac{X_{cooling}}{X_{in,power\ plant}} \tag{17}$$

$$\eta_x = \frac{X_{cooling}}{X_{in,power\ plant}+X_g+X_{crawl-space}} \tag{18}$$

$$\eta_{x,natural} = \frac{X_g+X_{crawl-space}}{X_{in,power\ plant}+X_g+X_{crawl-space}} = 1 - \frac{\eta_x}{\eta_{x,conventional}} \tag{19}$$

where $\eta_{x,conventional}$ is the conventional exergy efficiency, η_x is the exergy efficiency which takes into account the exergy supplied from the immediate natural exergy sources (one from the ground, X_g, and the other from the crawl-space, $X_{crawl-space}$ [W]), and $\eta_{x,natural}$ is the ratio of the exergy input from the immediate natural exergy resources to the total exergy input to the system.

4. Results and discussion

The main results of the analyses are presented in this chapter. Sensitivity of the results to the total efficiency including conversion efficiency of the power plant, distribution and transmission efficiencies of the grid (η_{TOT}), to the SFP of fans, and to the brine pump power can be found in Appendix B. Sensitivity of the system exergy performance to these parameters is presented in Appendix C.

4.1. Comparison of different space cooling systems without a cool exergy source

The chains of exergy flows from the initial natural gas input to the power plant to the environment are shown in Fig. 2 for Cases 1, 2, 3, 4, and 6. For the power plant, the exergy contained by natural gas is supplied to the power plant as fuel and the electricity produced is supplied to the heat pump. The difference between the exergy input from natural gas and the output electricity is the exergy consumption in the power plant. The same relationship between input, output and consumption applies also to the other components in the chain.

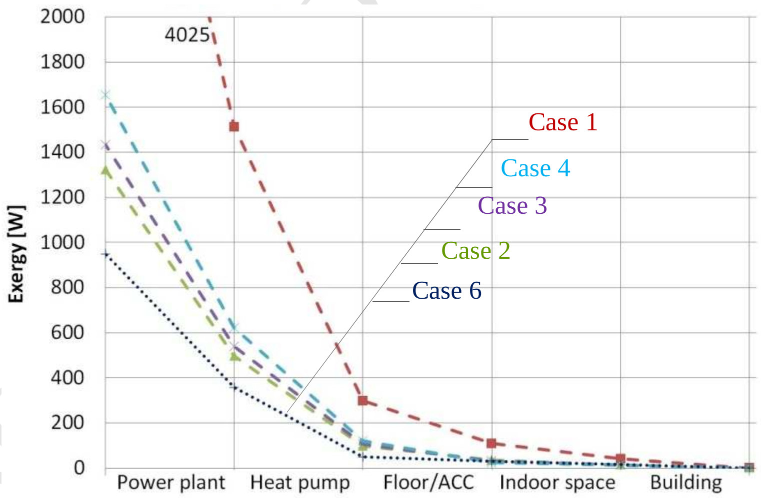
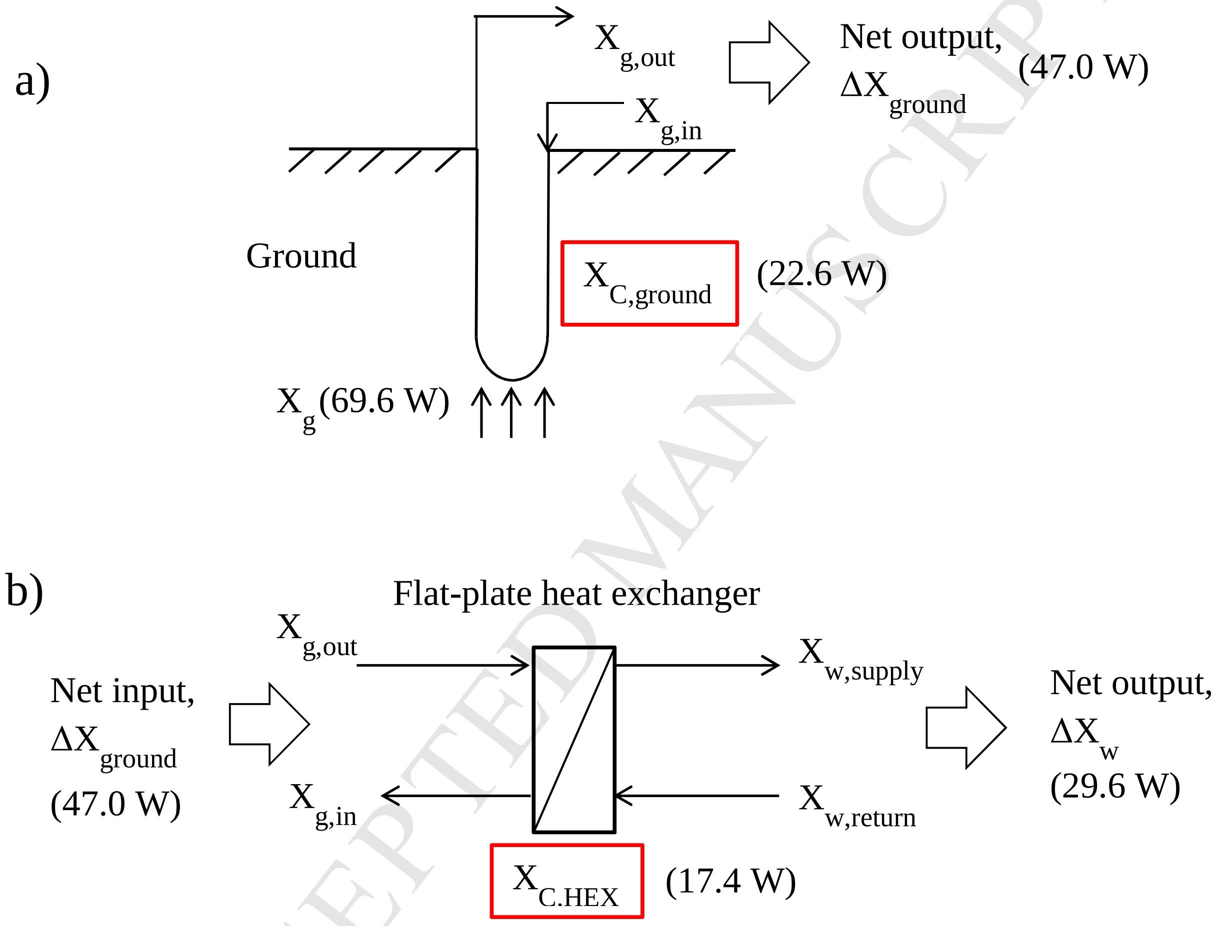


Fig. 2. Exergy flows for different cooling strategies without a cool exergy source (ACC: air-cooling coil)

ACCEPTED MANUSCRIPT

Exergy inputs, outputs, and consumptions in different system components are given in Fig. 3. $X_{g,in}$ and $X_{g,out}$ are the rates of exergy carried by the anti-freeze mixture flowing into and out of the ground heat exchanger, respectively [W].



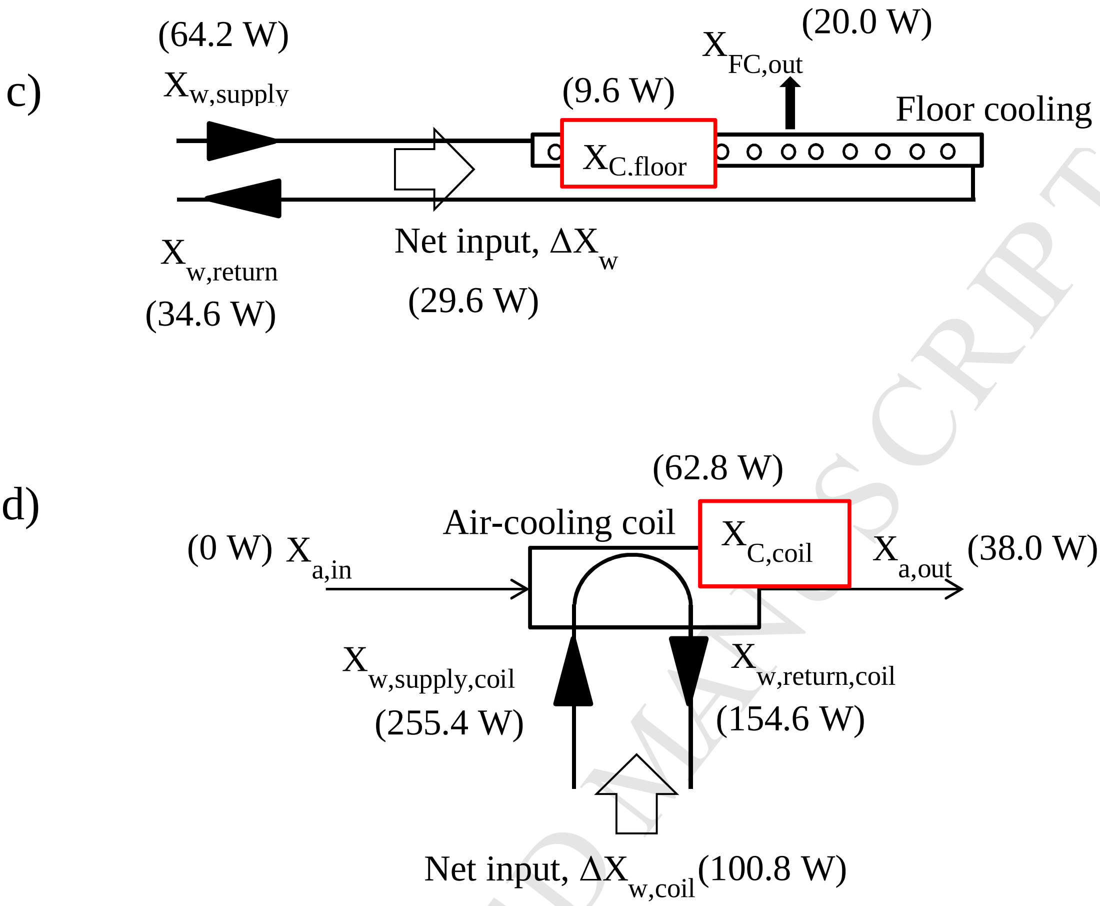


Fig. 3. Exergy input, output, and consumption in different system components: a) Ground b) Flat-plate heat exchanger c) Floor d) Air-cooling coil. a), b), and c) are given for Case 8. d) is given for Case 2. Values in the parentheses indicate the exergy values and the red boxes indicate exergy consumptions in the respective system components.

The exergy to be supplied to the power plant from natural gas is 4025 W for Case 1 and it is the largest among the investigated cases. This implies that the use of an internal solar shading device is not effective in reducing the cooling demand of the house. The large exergy input requirement is due to the large cooling load and also due to the way of addressing this load; cooling with an air-based system. Compared to the rest of the cases, exergy consumption in the rest of the system components is also the largest for Case 1. Case 1 clearly shows that it is crucial to minimize the cooling demand of the house.

In the rest of the cases in Fig. 2, an external solar shading device was employed. The exergy input required at the power plant has decreased remarkably compared to Case 1, and the exergy consumption in different system components has also decreased.

In Cases 2, 3 and 4, air cooling was employed to address the cooling load. Although the cooling exergy load itself is the same for all of these cases (13.9 W), the exergy required at the power plant in order to power the heat pump is different, due to different supply air temperatures assumed for each case (Table 3). The exergy consumption rate in the indoor space is 21.8 W, 16.2 W, and 10.7 W for Cases 2, 3 and 4, respectively. This trend is reversed for exergy consumption in the cooling coil, where the exergy consumption rates are 62.8 W, 76.0 W, and 96.7 W for Cases 2, 3 and 4, respectively. This is also reflected in the rest of the systems towards the source, where the differences between these cases in the heat pump and the power plant are clear in Fig. 2. Case 4 requires the highest exergy input among these three cases, followed by Case 3 and Case 2, therefore for the further analyses and comparisons, Case 2 will be used.

Among the cases presented in Fig. 2, Case 6 requires the lowest exergy input to the power plant, despite the cooling exergy load being slightly higher (15.8 W) than the air cooling cases. This is because of two reasons; the system being a water-based cooling system, and working at water temperatures close to room temperatures (high temperature cooling). The exergy input required at the power plant is 28% smaller for Case 6 (floor cooling) than Case 2 (air cooling).

4.2. The effects of immediate cool exergy sources on system performance

4.2.1. Cool exergy contained in the crawl-space

The crawl-space below the house acted as a buffer zone, where the air temperature was higher than the outdoor air temperature in the heating season and vice versa in the cooling season. This results in a warm or cool exergy storage effect in the crawl-space [12], [13]. The outdoor air temperature, air temperature in the crawl-space, and the specific exergy contained by air in the crawl-space are shown in Fig. 4.

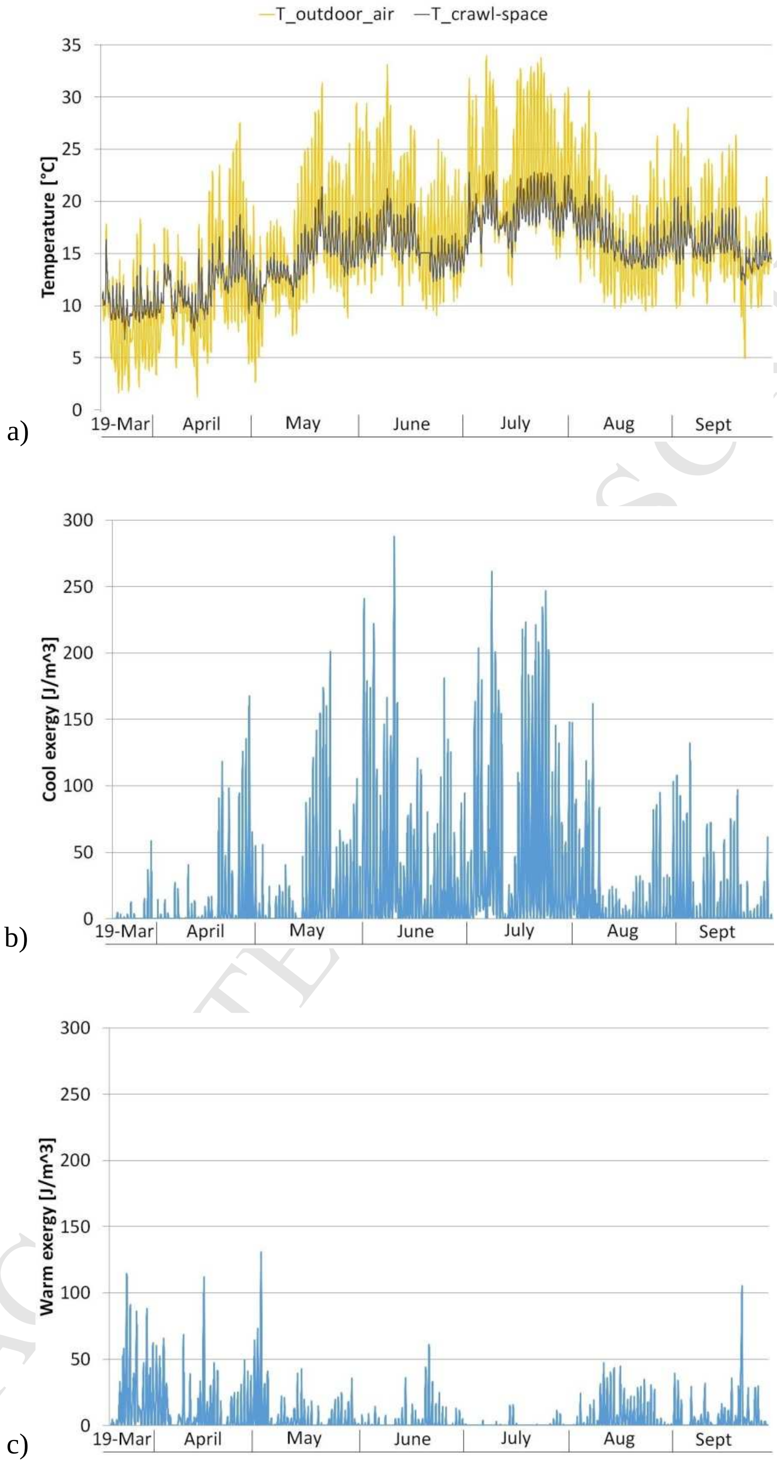


Fig. 4. a) Air temperatures outdoors and in the crawl-space b) Specific cool exergy contained by the air in the crawl-space c) Specific warm exergy contained by the air in the crawl-space

During the period from spring to autumn 2014, the maximum warm and cool exergy stored in the crawl-space were 131.1 J/m^3 and 287.9 J/m^3, respectively. Under the conditions considered in this study (air temperature in the crawl-space of 21.3°C and an outdoor air temperature of 30°C), the stored cool exergy density in the crawl-space corresponds to 152.7 J/m^3, which is lower than the maximum cool exergy density stored during this period. Fig. 4 shows that when the air temperature in the crawl-space is lower than the outdoor air temperature, there is cool exergy storage in the crawl-space.

Further cases were studied by modifying the boundary conditions of Case 6 and Case 2, in order to investigate the effects of this cool exergy storage on the whole system. That is, in Cases 7 and 5, it was assumed that the intake air was taken from the crawl-space, instead of the outdoor air. Fig. 5 shows the effects of the crawl-space (cool exergy storage) on the performance of different cooling systems.

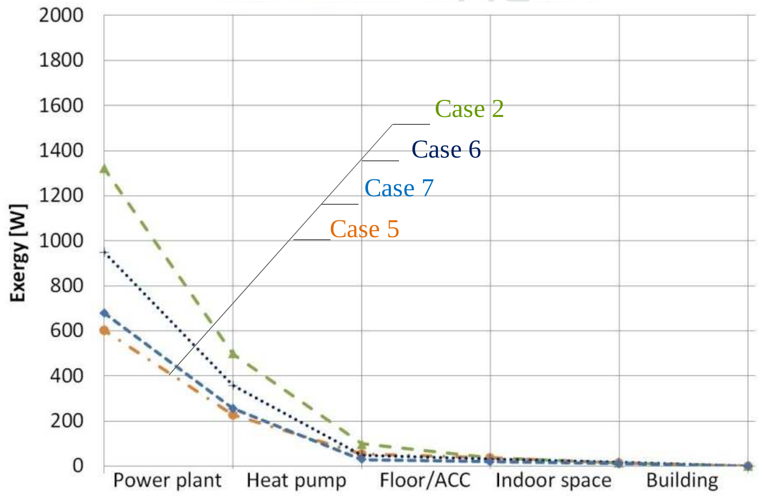


Fig. 5. Exergy flows for different cooling strategies with available cool exergy from the crawl-space (ACC: air-cooling coil)

In Fig. 5, the differences in the exergy input to the power plant between Case 2 and Case 5 (719 W, 54% reduction in exergy input to the power plant compared to Case 2), and between Case 6 and Case 7 (270 W, 29% reduction in exergy input to the power plant compared to Case 6) are due to the use of the cooler air at 21.3°C in the crawl-space instead of the outdoor air at 30°C as intake air. The rate of cool exergy provided from the crawl-space with the ventilation rates of 0.5 ach (Cases 6 and 7) and 1.2 ach (Cases 2 and 5) are 4.5 W and 10.7 W, respectively. The exergy consumption in the power plant, heat pump, floor, and in the indoor space is also decreased for Case 7 compared to Case 6. The exergy consumption in the power plant, heat pump, and cooling coil is decreased for Case 5 compared to Case 2.

It is worth noting that only 4.5 W and 10.7 W of cool exergy provided from the crawl-space can eliminate 270 and 719 W of exergy input by natural gas at the power plant, respectively. This is mainly because the actual cool exergy demand is very small so that making use of such small quantities of cool exergy results in a significant reduction on the supply side.

Fig. 5 shows that the air-based systems benefit more from the storage of cool exergy in the form of cooler air in a crawl-space compared to floor cooling. Although Case 5 requires less exergy input than Case 7, this does not mean that the air-based system performs better than the water-based system, due to the higher auxiliary energy use, as will be presented in 4.3. It should also be noted that compared to Case 7, Case 5 has higher exergy consumption in the cooling coil (19.0 W) compared to the floor structure (9.6 W), higher exergy consumption in the indoor space (21.8 W vs. 8.3 W), and has a higher cooling exergy load (13.9 W vs. 11.7 W).

4.2.2. Cool exergy contained in the ground

Although the floor cooling performs better than the air cooling cases as presented in Fig. 2, a closer look at the exergy flow reveals that it is possible to increase the exergy performance of this system with a better match of the exergy demand and supply. This is achieved through the coupling of the floor cooling system with a ground heat exchanger. The results are presented in Fig. 6.

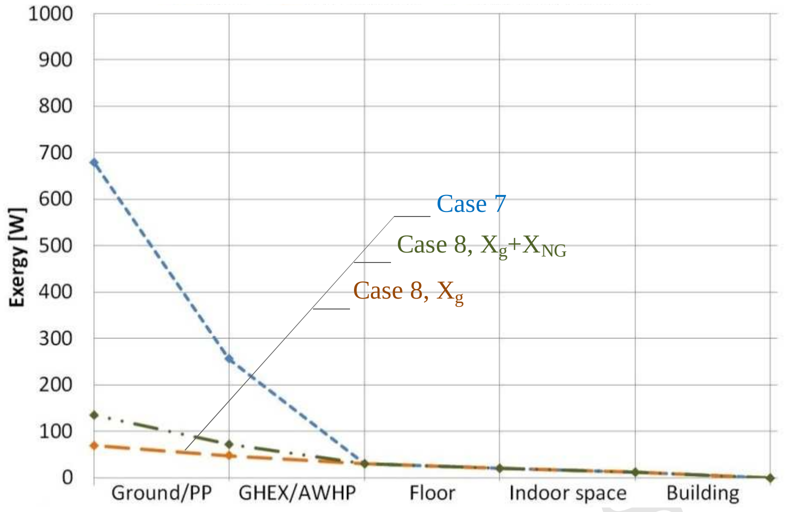


Fig. 6. Exergy flows for different cooling strategies with available cool exergy from the ground (PP: power plant)

The brine pump in the ground loop was assumed to be identical in terms of performance to the circulation pumps. The brine pump has a similar function to a heat pump, which transports the cool exergy from its source (ground) to where it is needed. This is also shown in Fig. 6, where the orange line (Case 8, X_g) is the flow of cool exergy from the ground through the system components, house and to the environment. The green line (Case 8, $X_g + X_{NG}$) shows, in addition to the cool exergy flow from the ground, the exergy input at the power plant from natural gas to provide the brine pump with electricity.

When comparing Case 7 to Case 8, the exergy input at the power plant is decreased from 680 W (for heat pump) to 65 W (for brine pump) corresponding to a 90% reduction. This difference is due to the use of the cool exergy available in the ground. When considering also the cool exergy that was initially available in the ground (69.6 W), then the total exergy input is 134.6 W, as given in Fig. 6 (Case 8, $X_g + X_{NG}$). Compared to Case 7, the use of ground without any thermodynamic refrigeration cycle (the use of stored cool exergy available in the ground) is an effective way to match the low exergy demand of the floor cooling system.

Regarding the exergy flow of the AWHP, a possible improvement could be to power it with on-site generated electricity from a renewable energy resource instead of a power plant in a remote location, though this requires further considerations for a definitive conclusion.

The pump power is important for realizing the true benefits of the cool exergy stored in the ground, as it is an addition to the cool exergy from the ground (cool exergy of 69.6 W from the ground is comparable to the 65 W of exergy by natural gas at the power plant to provide the brine pump with 24.5 W of electricity). This is crucial for justifying the free cooling. Increasing pump power requirements (e.g. to use a deeper or additional ground heat exchangers than one borehole or a worse pump) will decrease the overall efficiency of the system, as shown in Appendix C.

The initial design of the heating and cooling system of the house relied on the ground as the heat source and sink, and a theoretical single U-tube ground heat exchanger was designed. The benefits of using the ground compared to an AWHP were justified in energy terms in previous studies [23], [24] and the results obtained in this study justify this solution from the exergetic viewpoint.

Case 8 takes advantage of the cool exergy available both in the crawl-space (4.5 W) and in the ground (69.6 W). When comparing Case 6 to Case 8, the exergy input required at the power plant is decreased from 950 W (for heat pump) to 65 W (for brine pump) corresponding to a 93% decrease. This result emphasizes the benefits of using the naturally available heat sources and sinks in our immediate surroundings, although, a prerequisite of this is to limit the cooling (and heating) demand of the building as much as possible from the beginning of the design process. This implies that passive and active technologies should be well combined.

4.3. Auxiliary exergy input

In addition to the thermal exergy analyses, the effects of electricity inputs to pumps and fans on the whole exergy consumption patterns were also considered. The results calculated for all cases are presented in Fig. 7. In each case, there are four bars, which are, from left to right, the exergy input to pump, to fans, their total, and the exergy input to the power plant.

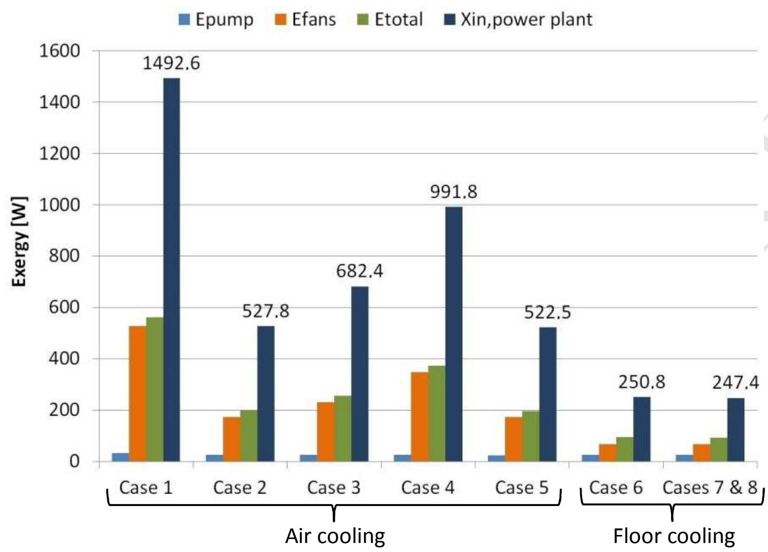


Fig. 7. Exergy inputs to the circulation pump, supply and exhaust fans and to the power plant

The results presented in Fig. 7 show that although the pump consumptions are within a close range, the fan consumptions vary greatly from floor cooling to air cooling cases. In the floor cooling cases, the ventilation system was only used to provide the necessary fresh air, while in air cooling cases, the ventilation system was used to remove all the necessary heat (cooling load), which resulted in larger ventilation rates. The large ventilation rates result in a relatively large auxiliary energy use compared to the floor cooling cases and this difference can be attributed to the difference between the air-based and water-based cooling approach. A high auxiliary energy use would decrease the energy and exergy-wise efficiency of the whole system (Appendix C).

Air-based systems require larger flow rates and volumes to transport the same amount of heat, cool or warm exergy compared to water-based systems because of the air’s smaller specific heat capacity and density than water. This causes a larger power requirement for air to be transported and it emphasizes the advantage of water-based heating and cooling systems.

Fig. 7 shows that the auxiliary exergy input to the system can be substantial compared to the thermal exergy values, as presented in Fig. 2, Fig. 5 and Fig. 6. These results emphasize the importance of minimizing the auxiliary component exergy use in order to achieve a holistically high performing system.

4.4.Exergy efficiency

In addition to the total exergy input to the systems, exergy efficiency can also be used to evaluate the overall exergetic performance of heating and cooling systems. The resulting exergy efficiencies for the studied cases are given in Table 5.

The exergy efficiencies given in Table 5 do not include the exergy input to the auxiliary components; the inclusion of the auxiliary components in the exergy efficiency will decrease the exergy efficiencies. Appendix C presents the exergy efficiency values when the auxiliary components are also considered. The effects of specific fan power, and brine pump power on overall system performance can also be found in Appendix C.

Table 5. Exergy efficiencies of different cases

	Case 1	Case 2	Case 3	Case 4	Case 5	Case 6	Case 7	Case 8
$\eta_{x,conventional}$ [%]	1.1	1.1	1.0	0.8	2.3	1.7	1.7	18.0
η_x [%]	1.1	1.1	1.0	0.8	2.3	1.7	1.7	8.4
$\eta_{x,natural}$ [%]	0	0	0	0	1.8	0	0.7	53.2

Exergy efficiency values show that the floor cooling (Case 6) has slightly higher exergy efficiency than the air cooling (Cases 1 to 4) when the effects of immediate natural exergy sources are not considered. When the effects of naturally available resources are considered, the exergy efficiencies increase as in Case 5 compared to Case 2,

ACCEPTED MANUSCRIPT

and Case 8 compared to Case 6. This is mainly due to the decreased exergy input into the power plant to provide the required space cooling, and through a better match of the exergy demand and supply (Case 8).

The natural exergy efficiency values ($\eta_{x,natural}$) clearly show that only Cases 5, 7 and 8 benefit from the available natural exergy sources in the immediate surroundings. In Case 8, 53.2% of the total exergy input to the system was provided by the natural exergy sources, whereas this value was 1.8% for Case 5 and 0.7% for Case 7. According to this definition, in case of a building using conventional heating and cooling systems, which does not use any immediate sources of natural exergy (as in Cases 1 to 4, and 6), $\eta_{x,natural}$ becomes zero. If in a building (e.g. a zero energy building), all the exergy requirement is supplied with solar, ground, wind, etc., this means that the exergy supply to the building is 100% sustainable and $\eta_{x,natural}$ becomes unity.

The results show that the heat sinks within our immediate environment should be used wisely, since these are valuable natural resources and they should not be exhausted through poor utilization policies. This is partly reflected in the exergy efficiency of Case 8; when the cool exergy consumption from the ground is taken into account then the exergy efficiency decreases to 8.4% from 18.0%. This implies that the rational efficiency index is η_x rather than $\eta_{x,conventional}$.

4.5. Overall discussion

This study considered a cooling season operation but it is crucial to assure that the given systems can work effectively in the heating season in residential buildings. Exergy analyses and especially the warm exergy concept can be used to analyze the performance of heating systems in buildings. The exergy performance of different heating systems has been addressed in another publication [15].

Certain assumptions have been made during the calculation procedure. The steady-state assumption brings certain limitations especially regarding the consideration of thermal storage effects and transient behavior of buildings. There are also certain limitations regarding the occupant presence and comfort throughout the day, since different control strategies (e.g. setback control) can be used in a single day. The current calculations do not address these issues, dynamic exergy calculations or exergy calculations based on building performance simulations (indoor environment and energy) can be used to study these effects.

The obtained results are dependent on the values used in the calculations (e.g. power plant efficiency, pump and fan powers, etc.) therefore it is crucial to choose realistic values for each parameter. In this study, the necessary values were obtained from literature or from the actual house components, when applicable. The uncertainties associated with these assumptions were examined and confirmed by sensitivity analysis, and the results are described in Appendix B.

For the air cooling cases, it was chosen to keep the same supply and return water temperatures from the heat pump and to vary the water and air flow rates, corresponding to a change in pump and fan powers. Another approach could have been to vary the supply and return water temperatures from the heat pump, and hence to improve the COP. Latent loads were not considered and it was assumed that the cooling in the air-cooling coil was sensible.

The effect of cooling system on thermal indoor environment was not directly studied; rather a comparison of exergy performance of different cooling systems was made based on the same level of comfort. The method used to condition spaces (chosen terminal units, air-based or water-based system) has a direct influence on the thermal comfort of the occupants. Human thermal comfort should not be sacrificed for energy- and exergy-wise efficiency improvement alone and they should be achieved simultaneously. Water-based radiant systems perform better compared to other systems in terms of occupant thermal comfort; they minimize the risk of unpleasant air movement (draught), and they create a uniform temperature distribution in spaces [27], [28].

The use of a crawl-space was applicable for this particular house but it might not always be applicable. The crawl-space has a similar function to an earth-tube dug into the ground, in fact an earth-tube also benefits from

ACCEPTED MANUSCRIPT

the cool exergy available in the ground. Radon should be considered for different locations, i.e. a radon barrier might be needed in locations where radon is a concern. In addition to radon, also the quality of air coming from the crawl-space can cause problems with the indoor air quality and it is crucial to consider this aspect in order not to cause any dissatisfaction to the occupants with the indoor environment. The moisture of the air coming in from the crawl-space can also be a problem, and the consideration of dehumidification needs may result in different exergy inputs and system performance. The moisture of the air from the crawl-space could cause problems regarding condensation on the floor cooling unless properly dehumidified. The condensation can be avoided through a dew point control on the water supply temperature to the floor cooling loops.

The usability of the heat sources and sinks that are found in our immediate environment (e.g. ground, lake, sea-water, etc.) will depend on the costs, regulations, and on geographical conditions. Nevertheless, the minimization of the cooling (and heating) demand is crucial since only with a reasonably low-exergy demand, it would be possible to use the naturally available exergy sources and sinks in our surroundings.

A high cooling load would limit the use of radiant systems (floor, ceiling or wall cooling) due to dew point concerns. The required low water temperatures might also limit the use of natural heat sinks and hence the use of a refrigeration cycle might become necessary.

If radiant systems are to be used with a high cooling load, then the air must be dehumidified, although this is not the optimal choice since it means that at another part of the system, water temperatures below the dew point are necessary to dehumidify the air, unless the air is dehumidified by other means (e.g. desiccant wheel). This operation strategy would partly off-set the benefits of high temperature cooling by the radiant floor cooling.

The optimal system design should be the one, in which the cooling demand is lowered as much as possible so that there is no need for dehumidification. Then high temperature cooling systems can be used which would enable the integration of heat sinks that can be found in our vicinity and this would increase the overall energy- and exergy-wise efficiencies.

5. Conclusion

The exergy performances of different space cooling systems were compared, using a single-family house as a case study. The main conclusions from the analyses are as follows.

1. Cooling exergy demand of the building should be minimized from the beginning to achieve reasonable exergy efficiency and also to allow the use of naturally available exergy sources and sinks. The cooling demand also influences the choice of indoor terminal units e.g. with a high cooling load, it might not be possible to use radiant cooling systems due to dew point concerns. The choice of the terminal unit is crucial since only with certain terminal units, it is possible to use natural resources, and since this choice directly affects the occupant thermal comfort.
2. The water-based radiant floor cooling system performed better than the air-based cooling with ventilation in terms of energy, exergy demand and consumption. When an air-to-water heat pump was used as the cooling source and the intake air was outside air, the exergy input required at the power plant was 28% smaller for the floor cooling system compared to the air cooling system. The water-based systems had remarkably smaller auxiliary exergy input compared to air-based systems, due to the use of water as the main heat carrier medium.
3. Cool exergy concept was used to quantify the available exergy in the crawl-space and in the ground. Integration of these natural exergy resources to the cooling system resulted in significant improvements in the system performance. The use of the cool exergy available in the crawl-space resulted in 54% and 29% smaller exergy input to the power plant for the air-based and water-based cooling systems, respectively. In these cases, only 4.5 W and 10.7 W of cool exergy provided from the crawl-space decreased the exergy input by natural gas to the power plant by 270 and 719 W, respectively.
4. The coupling of ground with the radiant floor cooling system is feasible since the exergy supply from the ground matches well with the low exergy demand of the floor cooling system. For floor cooling cases, it is possible to reduce the exergy input to the power plant by 90% and 93%, with the use of

ground, and use of the ground and the crawl-space, respectively. Brine pump power should be kept to a minimum to truly benefit from the “free” cooling through the cool exergy stored in the ground.

5. The benefit of coupling the ground with the floor cooling system was shown with the exergy efficiency values; 18% and 8.4% for the conventional and the new definition of the exergy efficiency, respectively, and 53.2% of the total exergy input to the system was from the ground. The decrease in efficiency indicates that the natural resources within our immediate surroundings should be used efficiently, and not exploited in ineffective ways through poor utilization.

It should be noted that the obtained results are case-specific, i.e. based on the house design, location, and steady-state assumption, and, therefore the results can differ as a function of these factors.

Acknowledgments

This study was financially supported by the Danish Energy Association’s Research and Development Program (ELFORSK), project no. 344-060, “Sustainable plus-energy houses”. The financial supports from DANVAK (Danish HVAC Society), Otto Mønsteds Fond, and Augustinus Fonden for an external research stay at the Tokyo City University are gratefully acknowledged. The authors thank Associate Prof. Dr. Jørn Toftum for his comments and help in editing the manuscript of this article.

References

[1] European Commission, "Directive 2010/31/EU of the European Parliament and of the Council of 19 May 2010 on the Energy Performance of Buildings," European Union, Brussels, 2010.

[2] ASHRAE - Presidential Ad-Hoc Committee on the Residential Construction Market, "Final Report - ASHRAE and the Residential Construction Market," American Society of Heating, Refrigerating and Air-

ACCEPTED MANUSCRIPT

Conditioning Engineers, Atlanta, 2014.

[3] P. Rohdin, A. Molin and B. Moshfegh, "Experiences from nine passive houses in Sweden - Indoor thermal environment and energy use," *Building and Environment,* no. 71, pp. 176-185, 2014.

[4] R. Holopainen, K. Salmi, E. Kähkönen, P. Pasanen and K. Reijula, "Primary energy performance and perceived indoor environment quality in Finnish low-energy and conventional houses," *Building and Environment,* no. 87, pp. 92-101, 2015.

[5] T. S. Larsen and R. L. Jensen, "Comparison of measured and calculated values for the indoor environment in one of the first Danish Passive Houses," in *Proceedings of Building Simulation 2011: 12th Conference of International Building Performance Simulation Association, Sydney, 14-16 November,* 2011.

[6] U. Janson, "Passive houses in Sweden - From design to evaluation of four demonstration projects," Lund University, Lund, 2010.

[7] M. Maivel, J. Kurnitski and T. Kalamees, "Field survey of overheating problems in Estonian apartment buildings," *Architectural Science Review,* vol. 58, no. 1, pp. 1-10, 2015.

[8] G. Sastry and P. Rumsey, "VAV vs. Radiant: Side-by-Side Comparison," *ASHRAE Journal,* pp. 16-24, 2014.

[9] E. Fabrizio, S. P. Corgnati, F. Causone and M. Filippi, "Numerical comparison between energy and comfort performances of radiant heating and cooling systems versus air systems," *HVAC&R Research,* pp. 692-708, 2012.

[10] T. Imanari, T. Omori and K. Bogaki, "Thermal comfort and energy consumption of the radiant ceiling panel

ACCEPTED MANUSCRIPT

ACCEPTED MANUSCRIPT

system: Comparison with the conventional all-air system," *Energy and Buildings,* pp. 167-175, 1999.

[11] B. Kilkis, "Role of radiant panel heating and cooling in net zero energy buildings," *ASHRAE Transactions,* pp. 602-609, 2011.

[12] R. Nishikawa and M. Shukuya, "Calculation of natural exergies," *Journal of Architectural Planning and Environmental Engineering, Architectural Institute of Japan,* pp. 39-46, 1998. (in Japanese).

[13] M. Shukuya, Exergy - Theory and Applications in the Built Environment, New York: Springer, 2013.

[14] R. Li, R. Ooka and M. Shukuya, "Theoretical analysis on ground source heat pump and air source heat pump systems by the concepts of cool and warm exergy," *Energy and Buildings,* pp. 447-455, 2014.

[15] O. B. Kazanci, M. Shukuya and B. W. Olesen, "Exergy performance of different space heating systems: A theoretical study," *Building and Environment 99,* pp. 119-129, 2016. doi: 10.1016/j.buildenv.2016.01.025.

[16] O. B. Kazanci and B. W. Olesen, "Sustainable Plus-energy Houses: Final Report," Technical University of Denmark, Department of Civil Engineering, Kgs. Lyngby, 2014.

[17] The Danish Ministry of Economic and Business Affairs, "The Building Regulations 2010," The Danish Ministry of Economic and Business Affairs, Danish Enterprise and Construction Authority, Copenhagen, 2010.

[18] H. E. Hansen, P. Kjerulf-Jensen and O. B. Stampe (Ed.s), Grundbog - Climatic systems, Copenhagen: DANVAK, 1997. (in Danish).

[19] J. Babiak, B. W. Olesen and D. Petráš, Low temperature heating and high temperature cooling, Brussels: REHVA - Federation of European Heating, Ventilation and Air Conditioning Associations, 2009.

[20] EN 1264-2, "Water based surface embedded heating and cooling systems - Part 2: Floor heating: Prove methods for determination of the thermal output using calculation and test methods," European Committee for Standardization, Brussels, 2008.

[21] EN 1264-3, "Water based surface embedded heating and cooling systems - Part 3: Dimensioning," European Committee for Standardization, Brussels, 2009.

[22] EN 1264-5, "Water based surface embedded heating and cooling systems - Part 5: Heating and cooling surfaces embedded in floors, ceilings and walls - Determination of the thermal output," European Committee for Standardization, Brussels, 2008.

[23] O. B. Kazanci, M. Skrupskelis, P. Sevela, G. K. Pavlov and B. W. Olesen, "Sustainable heating, cooling and ventilation of a plus-energy house via photovoltaic/thermal panels," *Energy and Buildings 83,* pp. 122-129, 2014.

[24] M. Skrupskelis and O. B. Kazanci, "Solar sustainable heating, cooling and ventilation of a net zero energy house," Technical University of Denmark, Kgs. Lyngby, 2012.

[25] EN 13779, "Ventilation for non-residential buildings - Performance requirements for ventilation and room-conditioning systems," European Committee for Standardization, Brussels, 2007.

[26] M. Shukuya, "Exergy concept and its application to the built environment," *Building and Environment,* pp. 1545-1550, 2009.

[27] K.-N. Rhee and K. W. Kim, "A 50 year review of basic and applied research in radiant heating and cooling systems for the built environment," *Building and Environment,* pp. 166-190, 2015.

ACCEPTED MANUSCRIPT

[28] B. Olesen, "Radiant Floor Cooling Systems," *ASHRAE Journal*, pp. 16-22, 2008.

Appendix A – Definitions of cool and warm exergy

Thermal exergy can be categorized into two types as "cool" and "warm" exergies and this approach enables us to properly consider the "warmth" or the "coolness" of a heat source or sink [12], [13]. An example is as follows: two tanks containing water are placed in an environment with an environmental temperature of T_o. One of the tanks is at a temperature of T_h and the other one is at a temperature of T_c, where $T_h>T_o$ and $T_c<T_o$. In the former case, the flow of energy and exergy are from the tank at T_h to the environment at T_o and this exergy corresponds to the flow of "warm" exergy, while in the other tank, the flow of energy is from the environment at T_o to the tank at T_c but the flow of exergy is from the tank to the environment and this flow of exergy is the "cool" exergy. It could be explained that, in cooling season, what is expected as a merit from this chilled water tank is the "cool" exergy and not the energy (the chilled water has a "lack" of energy). Further examples and more detailed descriptions of "cool" and "warm" exergies can be found in [13] and [14].

Appendix B – Sensitivity analysis

In order to quantify the effects of different assumptions on the results, sensitivity analyses were carried out on total efficiency including conversion efficiency of the power plant, distribution and transmission efficiencies of the grid (η_{TOT}), SFP of fans and on the brine pump power.

For the total efficiency (η_{TOT}), values of 0.3, 0.35 (used value), 0.4 and 0.45 were considered. Table B.1 shows the results. In Table B.1, the values indicate the exergy inputs to the respective system components for different cases.

ACCEPTED MANUSCRIPT

Table B.1. Exergy input to system components and to the power plant as a function of total efficiency

		Case 1	Case 2	Case 3	Case 4	Case 5	Case 6	Case 7	Case 8, X_g	Case 8, X_g+X_{NG}
Building [W]		42	14	14	14	14	16	12	12	12
Indoor space [W]		109	36	30	25	36	31	20	20	20
Floor/ACC [W]		300	98	106	121	55	49	30	30	30
Heat pump [W]		1515	498	539	622	227	358	256	47	72
Power plant [W]	**η_{TOT}=0.35***	4025	1323	1433	1654	604	950	680	70	135
	0.3	4695	1543	1672	1929	704	1109	793	70	146
	0.4	3522	1158	1254	1447	528	831	595	70	127
	0.45	3130	1029	1115	1286	469	739	529	70	120

*: Values presented for η_{TOT}=0.35 correspond to the values presented in Fig. 2, Fig. 5 and Fig. 6.

The results of the sensitivity analysis for the total efficiency show that although the absolute values of exergy input to the power plant change, the relative effects and relative system performances are the same.

Although not shown in Table B.1, the variation of power plant efficiency will also affect the necessary exergy input to the power plant for auxiliary components: a lower power plant efficiency results in a larger exergy input being necessary to supply the same power for the pumps and fans.

For the SFP of the fans, values of 1200 J/m^3 (used value), 1000 J/m^3, 750 J/m^3 and 500 J/m^3 were considered. These values are in SFP 3, SFP 2 and SFP 1 categories according to EN 13779:2007, respectively. The SFP values in Cases 6, 7 and 8 were not changed, since they use the measured values from the house [16]. The SFP values were only varied for the air cooling cases and the results are given in Table B.2. Pump powers were not changed; therefore the differences are only due to the variation of fan SFP.

Table B.2. Exergy input to the power plant for auxiliary components as a function of fan SFP

	SFP	Case 1	Case 2	Case 3	Case 4	Case 5	Case 6	Case 7 & 8
$X_{in,\ power\ plant,\ aux}$ [W]	1200 J/m^3*	1493	528	682	992	522	251	247
	1000 J/m^3	1259	451	580	838	446	251	247
	750 J/m^3	966	355	452	646	349	251	247
	500 J/m^3	674	259	324	454	253	251	247

*: Values presented for SFP=1200 J/m^3 correspond to the values presented in Fig. 7.

The results of the sensitivity analysis on fan performance show that even if the SFP were to be 500 J/m^3, the air-based systems cannot perform better than the water-based system in terms of exergy input required at the power plant for the auxiliary components. Even though the total exergy input required at the power plant becomes close for Cases 2 and 5 to water-based cooling systems' performance, it should be noted that water-based cooling cases used the actual measured values from the house in terms of fan energy use and therefore show the actual fan performance. Choosing better performing fans, as in this case with lower SFP values, will require other considerations, e.g. other products, costs, etc.

For the brine pump power, values of 10 W, 24.5 W (used value), 50 W and 75 W were considered. Table B.3 shows the effects of brine pump power on system performance.

ACCEPTED MANUSCRIPT

Table B.3. Sensitivity of the results to the brine pump power in Case 8

		X_g+X_{NG}			
	X_g*	**24.5 W***	**10 W**	**50 W**	**75 W**
Building [W]	12	12	12	12	12
Indoor space [W]	20	20	20	20	20
Floor [W]	30	30	30	30	30
GHEX [W]	47	72	57	97	122
Ground/power plant [W]	70	135	96	202	269

*: Values correspond to Fig. 6.

The results show that with an over-dimensioned pump, the whole system performance decreases; this indicates that in order to fully benefit from the available cool exergy in the ground, the dimensioning of the brine pump should be done carefully and the brine pump power should be minimized. An over-dimensioned pump will also mean reduced savings with the use of ground, compared to a system which uses an air-to-water heat pump, as in Fig. 6.

A sensitivity analysis on the power of the circulation pump was not carried out, since the characteristics of this pump were obtained from the actual component installed in the house. The pump has also a relatively small influence compared to the fan power.

The heat pump used in the calculations was not subject to a sensitivity analysis, because its performance was determined from tabulated data provided by the manufacturer as explained in 2.2.3. This allows the systems to be compared on a fair basis, using the same heat pump, and therefore reflecting the effects of actual operating conditions of air-based and water-based systems, e.g. the effects of water temperature leaving the evaporator of the heat pump.

It should be noted that heat pump plays a crucial role in the results and the effects of a given heat pump on the whole system performance should be considered for each case. The results presented in this study were obtained for one particular air-to-water heat pump.

The rest of the parameters used in the calculations either belonged to the actual design of the house or were actual components used in the house, therefore, further sensitivity analyses were not carried out.

Appendix C – Whole system exergy efficiency

Other than the total exergy input to the system (thermal exergy and for the auxiliary components), exergy efficiency can also be used to evaluate system performance. In addition to the efficiencies described based on thermal exergy values in 3.6, another exergy efficiency which takes the effects of auxiliary exergy use on the whole system performance into account was defined as follows

$$\eta_{x,system} = \frac{X_{cooling}}{X_{in,power\ plant} + X_{in,power\ plant,aux}} \tag{C.1}$$

where $\eta_{x,system}$ is the whole system exergy efficiency, and $X_{in,\ power\ plant,\ aux}$ is the exergy input to the power plant through natural gas to provide the necessary electricity to the auxiliary components [W] (calculated as defined in 3.5 and the values are given in Fig. 7).

Table C.1 shows the whole system efficiencies for different cases.

Table C.1. Whole system exergy efficiencies for different cases

	Case 1	Case 2	Case 3	Case 4	Case 5	Case 6	Case 7	Case 8
$\eta_{x,system}$ [%]	0.77	0.75	0.66	0.53	1.24	1.32	1.26	3.75
$\eta_{x,system}$' [%]*	0.9	0.88	0.79	0.66	1.63	1.44	1.42	5.57

*: SFP=500 J/m^3 for all systems, including the water-based cooling systems.

The results show that when the intake air is the outdoor air, water-based systems perform better than the air-based systems, i.e. they have a higher efficiency. The system performances are considerably affected by the availability of nearby natural heat sinks. When the cold air from the crawl-space is used as the intake air, the air-based system (Case 5) performs close to the water-based system coupled to a heat pump. When the water-based system is coupled to a ground heat exchanger (Case 8), it performs considerably better than all other system configurations.

If all systems have an SFP of 500 J/m^3, the overall performance of air-based systems increases compared to the previous cases. The water-based systems still perform better than the air-based systems when the intake air is the outside air.

When the intake air is from the crawl-space for the air-based system (Case 5), the air-based system performs better than the water-based system coupled to a heat pump; this is because the air-based system benefits from the available cool exergy in the crawl-space while the water-based systems either do not benefit from it (Case 6) or benefit in a limited amount compared to the air-based system (Case 7). These results also match with the results presented in 4.2.1. When the heat pump is replaced with a ground heat exchanger in water-based systems, water-based system performs considerably better than any other system configuration.

In order to examine the effects of brine pump power on the system exergy efficiency, two of the values given in Appendix B, 10 W and 75 W, were considered. With these values, the system exergy efficiency ($\eta_{x,system}$) turns out to be 4.28% and 2.62%, respectively. When an SFP of 500 J/m^3 is used ($\eta_{x,system}$'), the system exergy efficiency becomes 6.83% and 3.40%, respectively. Although these values are still considerably higher than the efficiencies of other systems, they also indicate that the brine pump power is a crucial parameter to be minimized, in order to properly and fully benefit from the available exergy in the ground.

The results show that the auxiliary exergy has a considerable effect on the overall system performance and therefore it should be considered carefully and minimized, in order to achieve an optimal system performance and to benefit from the naturally available heat sinks.

ACCEPTED MANUSCRIPT

Highlights

- Whole chains of exergy flows for different cooling systems were compared.
- The water-based floor cooling system performed better than the air-based cooling system.
- Cool exergy was used to study the effects of crawl-space and ground on system performance.
- A new exergy efficiency was used to show the benefits of coupling ground and floor cooling.
- Using cool exergy from natural resources results in significant exergy consumption reductions.

ACCEPTED MANUSCRIPT